

Dynamic performance assessment of a building-integrated cogeneration system for an Italian residential application



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ABSTRACT

This paper investigates the energetic, economic and environmental feasibility of a building-integrated micro-cogeneration system for an Italian residential application by means of transient simulations. The system under investigation was based on a natural gas-fuelled internal combustion engine cogeneration unit with 6.0 kW as rated nominal electric output. The plant was coupled with a multi-family house composed of three floors located in Naples (south of Italy), compliant with the transmittance values suggested by the Italian Law.

The analysis was carried out by means of a dynamic whole building simulation software; a detailed dynamic model was used to simulate the cogeneration device operation under both electric and heat load-following control strategies.

The simulation results were considered in order to compare the performance of the proposed system with a conventional system based on separate energy production. The simulations results showed that the proposed system makes it possible to reduce the primary energy consumption, the equivalent carbon dioxide emissions as well as the operating costs with respect to the reference system. In comparison to the electric load-following operation, the heat load-following control logic was more convenient from both energy and environmental point of views, but less advantageous from the point of view of operating costs.

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1. Introduction

The energy consumption of buildings significantly contributes to the overall European energy demand. Presently, this consumption is mainly due to the heating demand and is expected to grow significantly in the next years as a consequence of the significant increase of the air conditioning market [1]. Therefore the achievement of sustainability in the building sector necessitates a tremendous effort to reduce energy demand and boost energy efficiency of energy production systems. On the demand side, it is widely accepted that the improvement of the building envelope's insulation takes top priority; in order to limit the expected growth in building primary energy demand, the Italian government adopted a State Law [2] which enforces severe limits for walls and windows transmittances. Wide-ranging options exist on the supply side for the combined provision of home electricity and heating; among these options, micro-cogeneration (defined as the combined production of electrical and thermal energy from a single fuel source, with electric output not higher than 50 kW) is a

well-established technology and its deployment was considered by the European Community as one of the most effective measures to save primary energy as well as reduce greenhouse gas emissions [3]. The distributed generation nature of this technology also has the potential to reduce losses due to electrical transmission and distribution inefficiencies as well as alleviate utility peak demand problems.

Currently, micro-cogeneration devices, based on reciprocating internal combustion and Stirling engines, are already available in the market and a large R&D activity which aims at producing, in the medium and long term, small commercially available units based on fuel cells, gas and steam turbines, is already in progress [4]. However the estimation of the potential impact of cogeneration units is necessary to design policies as well as rank these systems among other potential energy saving and CO₂-reducing measures. Two main ways can be considered for studying and evaluating the performance of micro-cogeneration units under different load environments: (i) running field/lab experiments or (ii) developing accurate simulation models of micro-cogeneration devices. Taking into consideration that experimental analyses are both expensive and time consuming, several studies have investigated MCHP devices integrated into individual buildings using dynamic simulation tools able to capture transient behavior of the systems and compared their performance with other supply options.

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Nomenclature

Latin letters

A	area (m^2)
c	specific heat (kJ/kgK)
CU	unit cost (€/kWh , €/m^3)
E	energy (kJ)
GHG	greenhouse gas emissions
h	heat transfer coefficient ($\text{W/m}^2\text{K}$)
ICE	internal combustion engine
IEA	international energy agency
LHV	lower heating value (kJ/kg)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
MCHP	micro combined heat and power
OC	operating cost (€)
P	power (kW)
s	thickness (m)
SHC	solar heating and cooling
T	temperature ($^\circ\text{C}$)
U	transmittance (W/mK)
u	energy output-based emission factor
VAT	value added tax

Greeks

α	solar absorptance coefficient
Δ	difference
η	efficiency
λ	thermal conductivity (W/mK)
ρ	density (kg/m^3)

Superscripts

CS	conventional system
E_p	primary energy
E_{el}	electric energy
SP	proposed system

Subscripts

buy	bought from the central electric grid
B	boiler
CO_2	carbon dioxide
i	internal
e	external
el	electric
frame	frame of window
fuel	fuel
glazing	glazing of window
grid	national central grid
in	inlet
lim	limit value
MCHP	micro combined heat and power
max	maximum
out	outlet
p	primary
PP	power plant
sell	sold to the central electric grid
set	set-point
th	thermal
TOT	total
wall	wall
window	window
wd	window
x	generic pollutant

In the following, some of these studies are referred to, focusing on residential applications located in Italy. Di Pietra et al. [5,6] simulated the performance of an internal combustion engine-based MCHP unit (characterized by 35 kW as electric nominal output and 78 kW as nominal thermal output) integrated into two different multifamily houses located in four different Italian climatic zones. The simulations showed that, in comparison to a conventional system composed of a natural gas boiler and the central electric grid, the cogeneration system is able to reduce both primary energy consumption (the reduction was almost constant upon varying the city and equal to about 20%) and carbon dioxide equivalent emissions (mainly in cold climates). In [7] the analysis of the dynamic operation of a gas micro-turbine based trigeneration plant, integrated with the building hosting the University of Rome, was presented. A good agreement was found in comparing the simulation results derived by a computer tool developed by the Italian agency ENEA [8] integrating a mathematical model of the small-scale cogeneration device [9] with the experimental data. Arteconi et al. [10] developed a model to design, assess and optimize micro-generation plants from the energy, environmental and economic viewpoints. The model was tested in an Italian residential application, in order to compare different micro-cogeneration technologies. They found that, with respect to the conventional system, the micro-cogeneration device fuelled by natural gas allows for reducing both the primary energy consumption (from 10% up to 25%) and the annual CO_2 emissions (from 5% up to 20%). Caresana et al. [11] modeled an automotive-derived internal combustion engine cogeneration system (fuelled by natural gas and characterized by 28 kW as nominal electric output) with the aim to compare constant and variable speed operation modes; typical residential and service sector applications were chosen as test cases and the simulation results were discussed from an energy and economic point of views. The technical-economic analysis showed that good energy and economic performances compared with a separate generation system can be achieved both in household and in hotel applications, especially with variable speed operation. Cardona and Piacentino [12] developed a numerical model to simulate a fuel cell cogeneration system coupled with a thermal energy storage, in accordance with the energy requirements of a single-family residence. They found that the fuel cell cogeneration system is able to guarantee about 8% primary energy saving with regards to a conventional system composed of a natural gas furnace for space heating, a natural gas water heater, and an electric air conditioning unit for space cooling. Additional studies were performed in several other countries, such as Belgium [13], Germany [14], United Kingdom [15,16], Switzerland [17] and Canada [18,19]. All the above-mentioned works demonstrated that micro-cogeneration technology has the potential to simultaneously reduce the use of fossil fuels, the emissions of pollutants and the operating costs in comparison to other conventional supply systems.

In this paper the performance of a building-integrated micro-cogeneration system was simulated by using the whole-building simulation software TRNSYS [20]. The system analyzed basically consists of a micro-cogeneration device, an auxiliary gas boiler, a combined storage tank for both heating and domestic hot water production, pumps, thermostats, valves, diverters, fan-coils, and the control system. Simulations were performed for a multi-family house located in Naples (south of Italy) compliant with the Italian Standards in terms of the transmittance values of both the walls and windows [2]. An internal combustion engine-based micro-cogeneration device fuelled by natural gas was considered; this unit is commercialized by AISIN SEIKI company [21] and it is characterized by 6.0 kW as the nominal electrical output and 11.7 kW as the nominal thermal output. The detailed dynamic model [22,23] developed within Annex 42 of the International Energy Agency [24] was used to simulate the cogeneration device; this model was

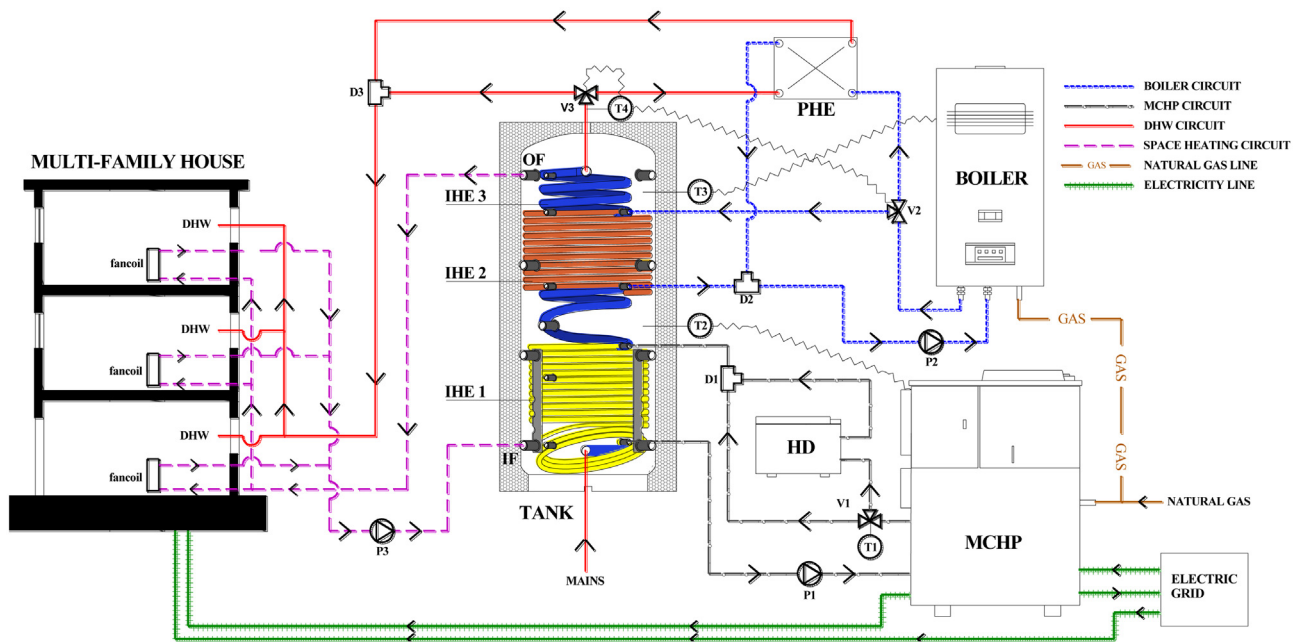


Fig. 1. Scheme of the system configuration analyzed in this study.

already calibrated and validated by the authors [25,26] based on a huge database obtained by means of several experiments carried out during both full and part-load MCHP operation. Plant performance was assessed by controlling the MCHP operation under both electric and heat load-following operations. Simulations were carried out for the heating period imposed by the Italian Law (from November 15th to March 31st) with a 1-min simulation time step.

The simulated performance of the proposed system was also compared with those of a conventional system composed of a natural gas fired boiler and the central electric grid. The comparison was performed in terms of primary energy consumption, carbon dioxide equivalent emissions and operating costs in order to assess the suitability of the proposed scheme in comparison to the reference system based on separate energy production.

The opportunity to use micro-cogeneration systems depends strongly on factors such as heat and power demand variations, control modes, the capacity and efficiency of the residential cogeneration system, as well as electricity import/export conditions and modes. Therefore the feasibility of a micro-cogeneration unit is a function of the design and size of the system as well as the building is intended for. For these reasons, studying and evaluating the performance of different systems under different load conditions is very important. The novelty of this work is mainly related to the fact that the dynamic performance of the AISIN SEIKI unit was never studied by means of a whole-building simulation software, and so its suitability was never accurately assessed. In some previous works [4,27,28] this cogeneration device was already analyzed by the authors only on the basis of some simplified hypotheses, such as steady-state operation and/or theoretical thermal/electric demand profiles with small variations in time around stepwise levels, an average unit cost for both electricity and natural gas fairly compliant with the Italian scenario, etc. However in a previously published paper [29] the authors already compared the experimental performance of the AISIN SEIKI unit with those derived by a simple steady-state evaluation based on nominal full-load efficiencies: this comparison showed that neglecting the influence of thermal transition implies a relevant overestimation of thermal energy supplied by the unit to the end-user (mainly during system operation at low electric levels) due to the inefficiencies associated to start-up and shut-down cycles. As a consequence, a dynamic approach is

mandatory due to the fact that static methods cannot be used to accurately assess the performance of the cogeneration device under investigation in this study. For these reasons in this work the system was ranked by fully considering the part-load characteristics of the cogeneration system; in addition the transient nature of building and occupant driven loads, as well as the updated Italian tariffs and incentives associated to both fuel consumption and electric energy purchased or fed to the grid were also taken into account.

The dynamic models selected for each component, boundary conditions, building envelope characteristics, building loads, as well as control logics are described in detail in the first part of the work; the assessment methodology, the performance indicators and results of the comparison between the proposed and conventional systems assumed as a reference are reported and commented in depth in the last sections of this paper.

2. Overall system description

Fig. 1 shows the scheme of the system configuration analyzed in this study. The whole plant basically consists of a micro-cogeneration (MCHP) device, a heat dissipator (HD), an auxiliary natural gas-fired boiler, a combined storage tank (TANK) for heating purposes and domestic hot water production with three internal heat exchangers (IHE1, IHE2, IHE3), a plate-fin heat exchanger (PHE), three pumps (P1, P2 and P3), four thermostats (T1, T2, T3 and T4), three three-way valves (V1, V2 and V3), three flow diverters (D1, D2 and D3), and fan-coils. Four different main circuits can be easily distinguished by analyzing Fig. 1: (1) micro-cogeneration device circuit, (2) natural gas-fired boiler circuit, (3) space heating circuit, and (4) domestic hot water circuit. Both the natural gas and electricity lines are also highlighted in Fig. 1.

The system under investigation is devoted to satisfying both the space heating load and domestic hot water requirements of a multi-family house (MFH), composed of three floors and located in Naples (South Italy). The heating purposes and domestic hot water production are satisfied by heating up the water contained in the storage. The MCHP system and natural gas-fired boiler co-operate in order to guarantee a given water temperature level (55 °C) within the tank. A group of fan-coils, supplied by the hot water storage, is

installed within the building in order to balance the sensible heating load. According to the European Standard EN12831:2003 [30], 20 °C was assumed as set-point indoor air temperature during the following time intervals:

- *week day*: from 7:30 to 11:00 and from 15:00 to 21:30;
- *weekend day*: from 8:30 to 11:00 and from 15:00 to 22:30.

Out of these periods, the indoor air temperature was not controlled.

Domestic hot water at 45 °C is produced by means of the internal heat exchanger IHE3 located in the tank; in the case of the temperature of the water exiting IHE3 is lower than 45 °C, the required additional heat is provided by the auxiliary natural gas-fired boiler.

According to the Italian Law [31], Italy is considered as divided in 6 different climatic zones (named A, B, C, D, E, F) based on the heating degree days index [32]. A maximum daily operation time for the system depending on the climatic zone where is located the building is defined in [31]: in the case of Naples (climatic zone C) the boiler can operate till a maximum of 10 h per day. In this study the boiler was allowed to operate for heating purposes only during the following time intervals:

- *week day*: from 7:30 to 11:00 and from 15:00 to 21:30;
- *weekend day*: from 8:30 to 11:00 and from 15:00 to 22:30.

The boiler was turned off out of these time intervals, except in the case thermal energy is required for producing hot water at 45 °C.

The Italian Law does not limit the daily operation time of cogeneration systems: as a consequence, in this study MCHP unit can operate till 24 h per day depending on the electric/thermal requests. System operation was investigated by considering the MCHP operation under both electric load-following logic and heat load-following control strategy. The generated electricity by cogeneration device is used directly in the building or is exported to the electric grid. The electric grid is also used to cover peak demand.

The components of the whole system were simulated using TRNSYS software [20], where each physical piece of the thermodynamic equipment is modeled with a component (named “Type”) that is a FORTRAN source code model. In the following Sections 2.1–3, the main features of the Types used in this study for simulating each plant component are described in detail.

The Italian Law [31] limits the time operation of heating systems during the winter season: according to [31], simulations were performed from November 15th to March 31st; 1-min simulation time step was used.

2.1. Hot water storage model

Storage tank was modeled by means of Type60f included in TRNSYS library; ten temperature levels (nodes) were used in the tank; a uniform tank loss coefficient per unit area equal to 3.0 kJ/hm²K was assumed. A vertical cylindrical liquid storage unit with one flow inlet (IF in Fig. 1) and one flow outlet (OF in Fig. 1) was considered: the flow inlet IF is used for receiving the cold water coming from the building, while hot water going toward the fan-coils located within the building exits the tank through the flow outlet OF. The tank was equipped with three internal heat exchangers: the hot water coming from the MCHP unit flows through the internal heat exchanger located in lower part of the tank (IHE1 in Fig. 1); the hot water coming from the natural gas-fired boiler goes toward the internal heat exchanger located in upper part of the tank (IHE2 in Fig. 1); the third internal heat exchanger (IHE3 in Fig. 1) allows heat to be extracted for domestic hot water production.

The use of thermal storage tanks prolongs the yearly operation time of a CHP facility and allows the cogeneration unit to

operate more continuously, avoiding frequent occurrence of transient behavior during start-up and shutdown [33]. Taking into consideration that the selection of the tank volume is quite important to optimize the plant performance, its optimal value was determined by carrying out a sensitivity analysis. The whole system operation with three different commercially available hot water storages was considered. The main characteristics of the three analyzed tanks are reported in Table 1 [34]: as it can be derived from this table, the TANK1 is characterized by a net volume (0.855 m³) about 1.2 larger than that one of TANK2 (0.738 m³) and about 1.7 times greater than that one corresponding to the TANK3 (0.503 m³); in addition it can be noticed that the surface area of the internal heat exchangers of the TANK1 is significantly higher than that one of the other two hot water storages, with a percentage difference ranging from 19% up to 28% with respect to the TANK2 and from 32% up to 50% in comparison to the TANK3; finally it can be highlighted that the height of thermostats controlling the operation of both MCHP and boiler is different among the three tanks under consideration.

The results of this sensitivity analysis are reported and discussed in detail in the Section 5.1.

2.2. MCHP model

An internal combustion engine-based micro-cogeneration unit commercialized by AISIN SEIKI company [21], fuelled by natural gas, with three cylinders, four strokes, and a total displacement of 952 cm³ was considered in this study. The manufacturer [21] provided the following nominal characteristics for this cogeneration device: 6.0 kW as nominal electrical output, 11.7 kW as nominal thermal output, 28.8% as rated electric efficiency at full load, and 56.2% as rated thermal efficiency at full load. This cogeneration device was modeled in this study by using the model [22,23] developed within the research annex (Annex 42) approved by the International Energy Agency [24]. The Annex 42 model is designed to predict primary power consumption, electric power generation, thermal output, coolant outlet temperature, etc. during both steady-state and transient operation and its instances were independently implemented into source code for four widely used building simulation tools (ESP-r [35], EnergyPlus [36], IDA-ICE [37], and TRNSYS [20]). This model was already calibrated and validated into TRNSYS environment by the authors [25,26] on the basis of several experimental tests.

The experiments showed no significant variations of its electric efficiency between transient and stationary regime; on the contrary, values of thermal efficiency during transient operation significantly lower than those measured during steady-state were derived by the measured data, highlighting the importance of considering the transient periods of the system in assessing its performance. The analysis of the tests performed on the AISIN SEIKI unit also indicated that the influence of both coolant temperature and coolant flow rate on both electrical and thermal efficiencies during steady-state operation can be considered as negligible, while they are strongly dependent from the electric output of the unit. In particular, the cogeneration device experimentally investigated showed that, whatever the cooling fluid temperature and flow rate are, during steady state operation the thermal efficiency increases (up to about 65%) at decreasing the electric output, while the electric efficiency rises (reaching a maximum value of about 26%) with increasing the electrical power. The comparison among simulation data and the experimental measurements highlighted the suitability of the model in predicting the cogeneration system performance: therefore annex 42 model (Type154) was used in this project for simulating the MCHP operation.

In this study the MCHP unit is connected to the hot water storage by means of the internal heat exchanger IHE1 (Fig. 1) and operates, together with the natural gas-fired boiler, in order to

Table 1

Main characteristics of the analyzed hot water tanks.

	TANK1	TANK2	TANK3
Tank volume (m ³)	0.855	0.738	0.503
Tank height (m)	2.040	1.750	1.780
Height of flow inlet (m)	0.125	0.125	0.107
Height of flow outlet (m)	1.691	1.401	1.453
Temperature levels used in the tank	10	10	10
Tank loss coefficient (kJ/hm ² K)	3.0	3.0	3.0
Destratification conductivity (kJ/hmK)	0	0	0
Number of internal heat exchanger	3	3	3
Height of thermostat controlling the MCHP (m)	0.816	0.700	0.712
Height of thermostat controlling the boiler (m)	1.836	1.575	1.602
The fraction of glycol contained in the IHE1 by volume (%)	0	0	0
IHE1 inside diameter (m)	0.027	0.026	0.026
IHE1 outside diameter (m)	0.029	0.028	0.028
IHE1 total surface area (m ²)	3.1	2.5	2
IHE1 length (m)	37.0	30.5	24.5
IHE1 wall conductivity (W/mK)	45	45	45
IHE1 material conductivity (W/mK)	45	45	45
Height of IHE1 inlet (m)	0.813	0.622	0.715
Height of IHE1 outlet (m)	0.144	0.138	0.120
The fraction of glycol contained in the IHE2 by volume (%)	0	0	0
IHE2 inside diameter (m)	0.026	0.026	0.026
IHE2 outside diameter (m)	0.028	0.028	0.028
IHE2 total surface area (m ²)	2.5	1.8	1.25
IHE2 length (m)	30.5	21.9	15.6
IHE2 wall conductivity (W/mK)	45	45	45
IHE2 material conductivity (W/mK)	45	45	45
Height of IHE2 inlet (m)	1.521	1.162	1.221
Height of IHE2 outlet (m)	1.048	0.848	0.920
The fraction of glycol contained in the IHE3 by volume (%)	0	0	0
IHE3 inside diameter (m)	0.023	0.023	0.023
IHE3 outside diameter (m)	0.025	0.025	0.025
IHE3 total surface area (m ²)	7.8	5.8	5.3
IHE3 length (m)	106.5	80.2	72.1
IHE3 wall conductivity (W/mK)	16	16	16
IHE3 material conductivity (W/mK)	16	16	16
Height of IHE3 inlet (m)	0.108	0.108	0.090
Height of IHE3 outlet (m)	1.703	1.418	1.470

heat up the water within the tank. Thermal power during unit operation is recovered from the exhaust gases and the engine jacket of the micro-cogenerator by means of a water–glycol mixture and transferred to the water within the storage tank by using water as coolant. As suggested by manufacturer [21], 33.5 l/min was assumed as coolant flow rate.

Both electric load-following MCHP logic and heat load-following MCHP control strategy were evaluated:

- In the case of electric load-following operation, the cogeneration system produces electricity by modulating the power supplied through the inverter on the basis of the electric demand. Heat is also available during unit operation, even if its level depends on the system electric output. The electric grid is used when the end-user electric request becomes larger than the maximum MCHP electric output. In the case of electric load is not required, the unit is turned off.
- In the case of thermal load-following operation, the system operates according to the external temperature signal coming from the thermostat T2 placed on the tank near the exit of the internal heat exchanger IHE1: when this temperature is lower than the set-point value (55 °C), the unit operates providing its maximum electric and thermal outputs; when the water temperature in the storage exceeds the set-point value, the unit is turned off. Any unused excess of current that is not self-used is sent to the power line.

The AISIN SEIKI unit has an internal control to keep the internal water-glycol temperature lower than a given value (about 70 °C), in order to prevent conditions endangering the engine block. This

particular behavior of the cogeneration device was also taken into account in this study by means of the heat dissipator (HD) that operates in the case of the temperature of hot water coming from the cogeneration device (sensed by the thermostat T1 in Fig. 1) becomes higher than 70 °C.

2.3. Boiler model

The natural gas-fired boiler was modeled in TRNSYS by using the Type6 included in TRNSYS library. It can interact with the water within the tank by means of the internal heat exchanger IHE2.

In this study the auxiliary heater is activated only when the water temperature within the tank (sensed by the thermostat T3 in Fig. 1) is lower than a given set-point (55 °C) or in the case of the domestic hot water temperature shows a temperature (sensed by the thermostat T4 in Fig. 1) lower than 45 °C. Cold water coming from the mains is heated up during the flow through the third internal heat exchanger located in the tank (IHE3 in Fig. 1) and supplied to the end-user at 45 °C. Domestic water temperature exiting the tank is sensed by thermostat T4 (Fig. 1); this thermostat controls the operation of both three-way valves V2 and V3: in the case of temperature of water exiting IHE3 is lower than 45 °C, both hot water coming from the boiler and domestic water coming from the tank are redirected toward the plate heat exchanger PHE (Fig. 1). Heat is transferred from the hot water in the boiler circuit flow to mains domestic water through this heat exchanger, thereby providing hot water to the taps at 45 °C. So in the case of temperature TT4 measured by thermostat T4 is lower than 45 °C, demand from the central heating is bypassed allowing the full boiler output to be utilized for hot water; otherwise the three-port valve V2 redirects

the boiler flow back to its usual route. The plate heat exchanger PHE was modeled by using the Type5b (included in TRNSYS library) with an overall heat transfer coefficient equal to 3780 kJ/hK.

Depending on the control function value (zero or one) coming from the two thermostats T3 and T4, the Type6 performs like a furnace adding heat at a maximum rate $P_{th,B,max}^{SP}$, without exceeding a maximum water outlet temperature $T_{out,B,max}^{SP}$ (that may be thought of as an arbitrary safety limit).

A boiler with $P_{th,B,max}^{SP} = 20.0$ kW and $T_{out,B,max}^{SP} = 85$ °C [38] was considered for the analysis carried out in this study. The heat flow rate provided by the boiler is calculated as follows:

$$P_{th,B}^{SP} = \dot{m}_{w,B}^{SP} \cdot c_w \cdot (T_{out,B,max}^{SP} - T_{in,B}^{SP}) \quad (1)$$

where $\dot{m}_{w,B}^{SP}$ is the water mass flow rate going through the boiler, c_w is the water specific heat (assumed equal to 4.190 kJ/kgK) and $T_{in,B}^{SP}$ is the water temperature entering the boiler. According to the manufacturer data [38], 9.5 l/min as water flow rate was considered in this study. In the case of $P_{th,B}^{SP}$ is higher than $P_{th,B,max}^{SP}$, the Type6 considers $P_{th,B}^{SP} = P_{th,B,max}^{SP}$, and the temperature of the water exiting the boiler is calculated as follows:

$$T_{out,B}^{SP} = T_{in,B}^{SP} + \frac{P_{th,B,max}^{SP}}{\dot{m}_{w,B}^{SP} \cdot c_w} \quad (2)$$

The boiler manufacturer [38] suggests 0.924 as boiler efficiency for a part load ratio $P_{th,B}^{SP}/P_{th,B,max}$ equal to 30% and 0.927 as boiler efficiency in the case of operation at rated output. In this study the boiler efficiency η_B^{SP} is evaluated by considering a linear interpolation of the manufacturer data:

$$\eta_B^{SP} = 0.924 + 0.000214 \cdot (P_{th,B}^{SP} - 6.0) \quad (3)$$

Exhaust and combustion losses rates are considered as negligible in this study: therefore the primary power consumption of the boiler $P_{p,B}$ is evaluated as follows:

$$P_{p,B}^{SP} = \frac{P_{th,B}^{SP}}{\eta_B^{SP}} \quad (4)$$

2.4. Building data and loads

The Type56a (included in TRNSYS library) was used for modeling the building envelope, indoor air set-point temperature, infiltration and internal gains. A detailed description of both building envelope and loads can be found in the following sections.

2.4.1. Building envelope characteristics

A multi-family house composed of three floors located in Naples (latitude: 40° 50' N; longitude: 14° 15' E) was modeled in this study. The geometric layout of the MFH is basically a multiplication of a single family house (SFH) type building geometry. The main geometrical characteristics of each floor and the building orientation are shown in Fig. 2. All floors have the same useable floor area (96.0 m²), while the net height of each floor is 3.0 m. Five windows were considered for each floor.

Type15-6 (included in TRNSYS library) was used for reading the external EnergyPlus weather data file [39] of Naples.

Starting from the 1st January 2010, the Italian Law [2] specifies the threshold values of transmittance for both walls and windows of renovated buildings depending on (i) the climatic zone where the building is located and (ii) the wall type (external wall, ground, roof). These threshold values of transmittance U_{lim} for property renovations located in Naples are specified for walls and windows, respectively, in Tables 2 and 3.

In Table 2 the characteristics of building walls considered in this study are also specified: in particular, internal (α_i) and external

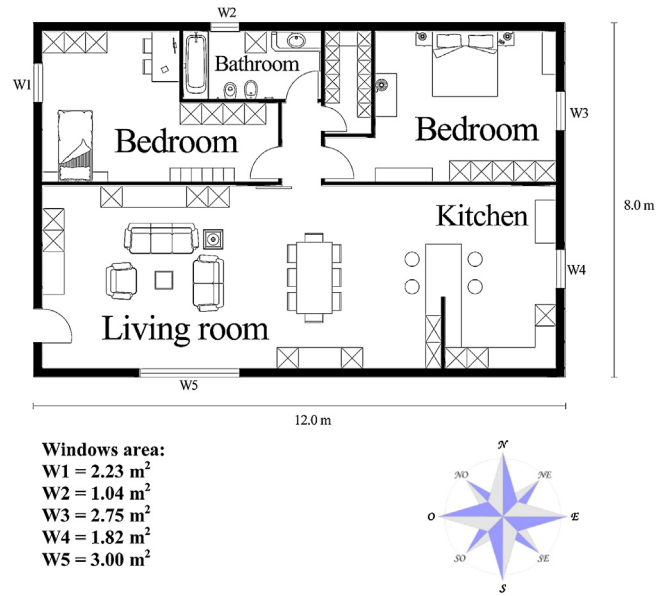


Fig. 2. Main geometrical characteristics of a single flat of the whole multi-family house.

(α_e) solar absorptance, internal (h_i) and external (h_e) convective and radiative heat transfer coefficients of walls, wall transmittance (U_{wall}), layers material, thickness (s), density (ρ), specific heat (c) and thermal conductivity (λ) are reported for each wall type. As it can be derived from Table 2, each wall is compliant to the requirements of Italian Law [2], with transmittance values equated to the given threshold values; values of h_i and h_e were derived by the European Standard EN ISO 6946 [40], while the values of ρ , c and λ were defined based on Italian Standard UNI 10351:1994/EC [41]. The characteristics of the external walls and the ground reported in Table 2 are the same for all flats; the characteristics of the roof specified in the same table are related to the roof of the whole building.

The main characteristics of building windows are reported in Table 3: in particular, internal (h_i) and external (h_e) convective and radiative heat transfer coefficients of windows, frame transmittance U_{frame} , glazing transmittance $U_{glazing}$, window type, glazing width, spacing width, spacing gas, ratio between frame area A_{frame} and window (glazing + frame) area A_{window} are specified. As it can be derived from Table 3, windows transmittance value is compliant to the requirements of Italian Law [2] with transmittance value equated to the given threshold value; values of U_{frame} and U_{window} were calculated based on the European Standard EN ISO 10077-1 [42]; all windows are provided with shadings, whose zone and wall reflection coefficients were considered equal to 0.5. According to the Italian Law [43], the area of each window was defined as the 12.5% of the ground area of the room where the window is located.

2.4.2. Internal gains and infiltration

Heat coming from occupants, personal computers and lighting systems was assumed to contribute to the internal gains. Fig. 3 shows the number of occupants of each single floor and the occupants-related sensible heat gain of each single floor as a function of the time during a week day (Fig. 3a) and during a week-end day (Fig. 3b). Sensible heat coming from each occupant was assumed equal to 75 W: this values is suggested by the Standard ISO 7730 [44] in the case of light work/typing as degree of activity.

The operation of one personal computer (providing 140 W as sensible heat) was also considered from 14:00 to 15:30 and from 21:00 to 22:30; no difference between week day and weekend was considered.

Table 2
Characteristics of building walls.

Wall type	U_{lim} (W/m ² K)	α_i	α_e	h_i (W/m ² K)	h_e (W/m ² K)	U_{wall} (W/m ² K)	Layer materials	s (m)	ρ (kg/m ³)	c (kJ/kgK)	λ (W/mK)
External walls of all flats	0.40	0.75	0.30	7.7	25.0	0.40	Gypsum plaster	0.02	900	1.00	0.250
							Bricks	0.33	1800	0.84	0.720
							Expanded polyester	0.07	20	1.20	0.041
							Gypsum plaster	0.02	900	1.00	0.250
Roof of the building	0.38	0.35	0.75	7.7	25.0	0.38	Gypsum plaster	0.015	1800	1.00	0.700
							Cement and bricks	0.300	1000	0.84	0.300
							Concrete	0.084	1900	0.84	1.060
							Bitumen	0.010	1200	1.00	0.170
							Expanded polyester	0.050	20	1.20	0.041
							Sand	0.050	1700	0.84	1.400
							Lime and cement adhesive	0.010	1500	1.00	0.720
							Tiles	0.010	2300	0.84	0.720
Ground of all flats	0.42	0.60	0.60	7.7	25.0	0.42	Tiles	0.010	2300	0.84	0.720
							Lime and cement adhesive	0.010	1500	1.00	0.720
							Sand	0.050	1700	0.84	1.400
							Expanded polyester	0.040	20	1.20	0.041
							Bitumen	0.005	1200	1.00	0.170
							Concrete	0.077	1900	0.84	1.060
							Bricks	0.300	1000	0.84	0.300
							Gypsum plaster	0.015	1800	1.00	0.700

Table 3
Characteristics of windows.

U_{lim} (W/m ² K)	h_i (W/m ² K)	h_e (W/m ² K)	U_{frame} (W/m ² K)	$U_{glazing}$ (W/m ² K)	Window type	Glazing width (mm)	Spacing width (mm)	Spacing gas	A_{frame}/A_{window} (%)
2.6	7.7	25.0	2.94	2.54	Double glazing	6	16	Argon	15

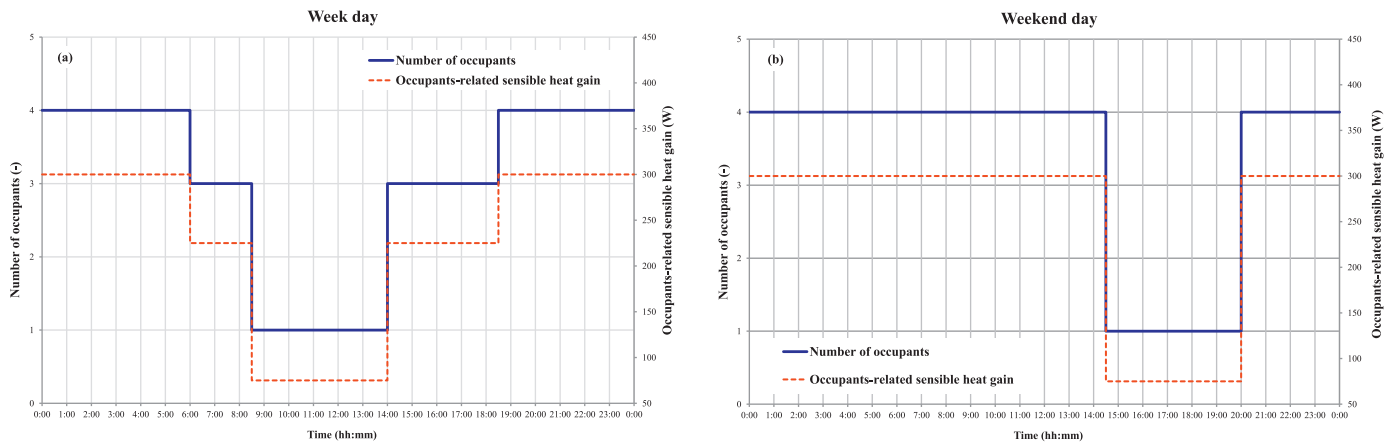


Fig. 3. Occupants-related sensible heat gains for a single flat during week day (a) and weekend day (b).

Fig. 4 shows the sensible heat flux related to the lighting appliances installed in the rooms of each single flat. The thermal power coming from each lighting system was assumed equal to the 75% of its nominal electric capacity. Heat produced by lighting systems was considered as transferred to the indoor air by both radiation (70%) and convection (30%).

Concerning the infiltration, the European Standard EN 12831:2003 [30] was used to calculate the infiltration air flow rate induced by wind and stack effect on the building envelope: based on this European Standard, a natural air change equal to 0.28 was considered.

The internal gains coming from occupants, personal computers and lighting appliances for the entire building were generated by superposition of three single flat profiles.

2.4.3. Space heating load

A group of fan-coils, supplied by the hot water storage, is installed within the building in order to balance the space heating load. A temperature difference of 5 °C was assumed for the hot water crossing the fan-coils, with a variable-flow pump controlling the fan-coils outlet temperature.

Fig. 5a shows the daily building thermal load profiles for four typical weekend days (November 30th, December 30th, January 31st and March 15th): the values of thermal load are different from zero only from 7:30 to 11:00 and from 15:00 to 21:30 due to the fact that the indoor air temperature was controlled (target is fixed at 20 °C) only during these time intervals; in the remaining time the thermal load is zero because the indoor air temperature was not controlled.

Fig. 5b reports the load-duration diagram with the heat-demand values sorted in descending order. As it can be derived from Fig. 5b, space heating demand has a duration of about 1350 h during the whole heating season (space heating is not required during the remaining time). In addition it can be highlighted that the building sensible thermal load is lower than the nominal thermal output of the MCHP unit (11.7 kW) for the most part of the system operation time. This maximum thermal load was set in TRNSYS simulations as the sum of the thermal output of the boiler (20.0 kW) and the maximum thermal output of the MCHP unit (11.7 kW).

The values of both thermal energy required for heating purposes and thermal energy required for heating purposes per unit reference floor area derived from simulations are specified in Table 4;

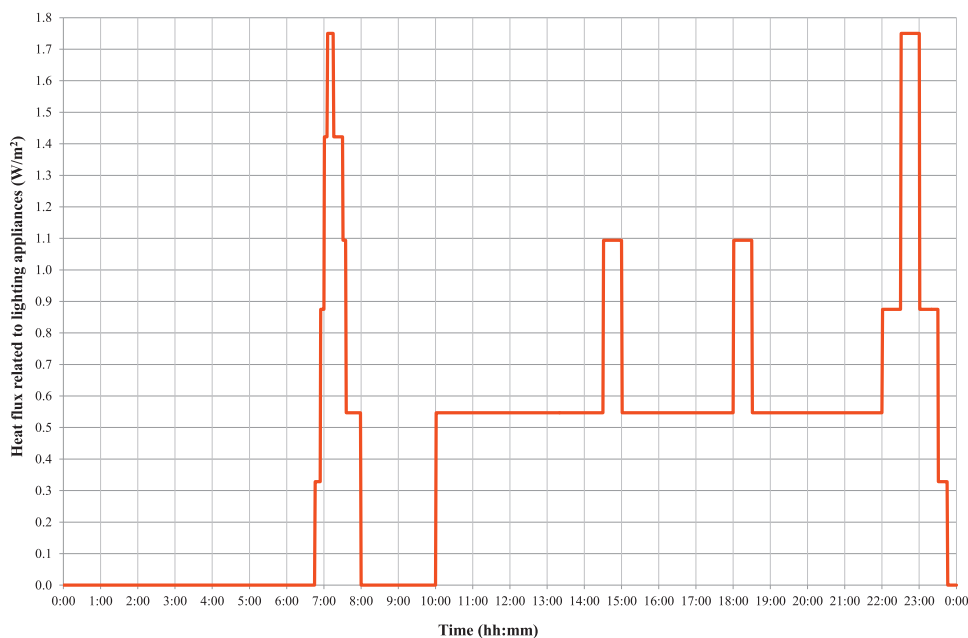


Fig. 4. Daily heat flux related to the lighting appliances for a single flat.

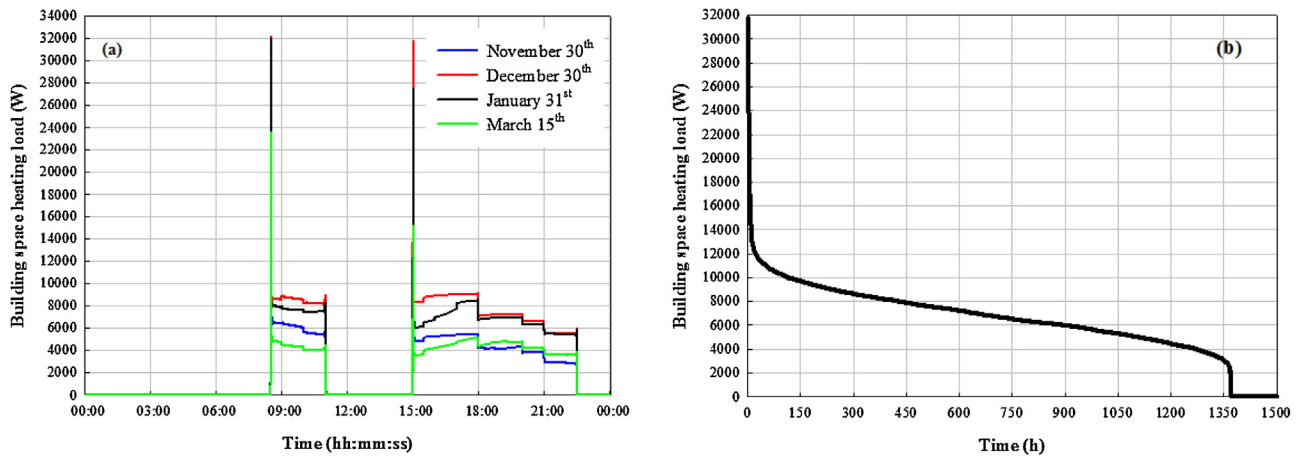


Fig. 5. Daily building space heating load profile for four typical days (a) and space heating load-duration diagram (b).

data were reported for each month and for the entire heating season.

2.4.4. Domestic hot water demand profile

Sets of yearly load profiles for the domestic hot water demand in the time scales of 1 min, 6 min, and 1 h were specified within IEA-SHC Task 26 [45]. Each profile consists of a value of water flow rate for every time step. Each set of load profiles contains profiles with different average basic load (100, 200, 400, 800 l/day) and different initial random values.

In this study, the domestic hot water demand profile with an average basic load of 200 l/day in the time scale of 1 min issued by IEA-SHC Task 26 [45] was used for each single flat. Fig. 6a shows the daily domestic hot water demand of each single flat for four typical days (November 30th, December 30th, January 31st and March 15th), while Fig. 6b reports the hot water demand-duration diagram of each single flat with the domestic hot water flow rate values sorted in descending order. As it can be derived from Fig. 6b, domestic hot water is required from each single flat for about 105 h during the whole heating season (no domestic hot water is required during the remaining time), with a maximum flow rate of about 1300 l/h. In this paper the domestic hot water demand profile of the entire building was generated by superposition of three of the single flat profile reported in Fig. 6a.

The temperature of cold water coming from the mains was derived by considering the distribution profile defined by the Type15-6 (included in TRNSYS library).

2.4.5. Electricity demand profile

Fig. 7 highlights the daily electric demand profile for each single flat resulting from the operation of both lighting systems and other domestic appliances (vacuum cleaner, dishwasher, washing machine, PC, TVs, fridge). The electric consumptions of each appliance assumed in this work were derived from [46]. No difference was considered among week days and weekend days. The daily electric demand profile of the entire building (3 flats) was defined by superposition of the profile reported in Fig. 7. The electric demand profile considered in this study corresponds to an electric consumption for the whole building of about 109.7 Wh/m² per day.

Table 4

Thermal energy required by the whole building for heating purposes.

	December	January	February	March	Whole season
Thermal energy consumption (kWh)	1900	2395	2017	2106	9631
Thermal energy consumption per unit area (kWh/m ²)	6.6	8.3	7.0	7.3	33.4

2.4.6. Further energy demands

Three circulation pumps are included in the simulation model:

- (1) pump P1 for heat recovery from the MCHP unit (Fig. 1);
- (2) pump P2 for moving the water within the boiler loop (Fig. 1);
- (3) pump P3 for the heat distribution within the building (Fig. 1).

Both the pumps P1 and P2 were modeled by using the Type114 (included in TRNSYS library) as single (constant) speed pumps, while the pump P3 was simulated by means of Type110 (included in TRNSYS library) by assuming the fraction of the motor heat loss transferred to the fluid stream as negligible.

The electricity demand of the pumps were taken into account: the electric power consumption during operation was assumed equal to 0.75 kW for the pump P3, and 0.55 kW for both the pumps P1 and P2.

The electric consumption (130 W) of the fan of the heat dissipator used for dissipating the excess heat recovered by the MCHP unit was also considered; the electrical consumption (238 W) of fan coils installed within the building was also taken into account.

During both stand-by and cool-down operation, the cogeneration unit does not supply electric energy to the end-user and the electronic controllers within the unit require some electricity: the Annex 42 model allows for accounting the electric consumption of the MCHP device during both stand-by (90 W) and cool-down operation (190 W).

3. MCHP and boiler control logics

As above-mentioned, in this study the MCHP operation is controlled by the thermostat T2 (Fig. 1), while the boiler operation is controlled by both the thermostats T3 and T4. In this TRNSYS project both MCHP loop and boiler loop are equipped with an on/off differential controller modeled by Type2d (included in TRNSYS library).

In this study 55 °C was considered as set-point temperature for both thermostat T2 and thermostat T3; 2 °C was assumed as upper dead band ΔT_h for both MCHP circuit and boiler circuit; 2 °C was considered as lower dead band ΔT_l for MCHP loop, while ΔT_l was considered equal to 0 °C for the boiler loop.

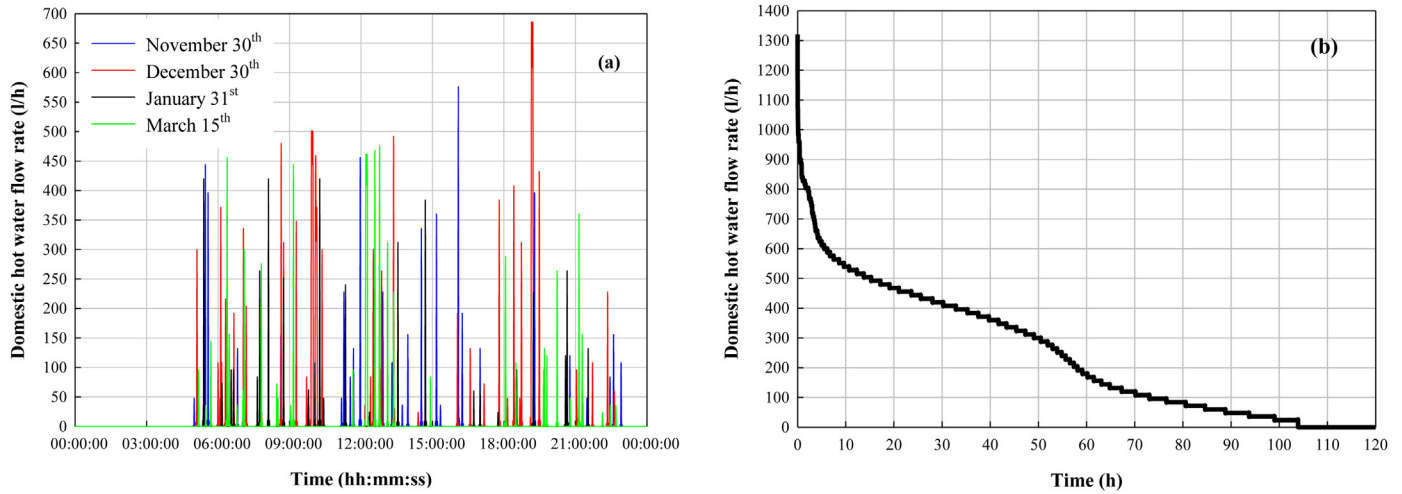


Fig. 6. Daily domestic hot water demand profile for a single flat during 4 typical days (a) and domestic hot water duration curve for a single flat (b).

4. Reference system

The main focus of this study is comparing the performance of the proposed plant based on the cogeneration device with those of a conventional system (CS) based on separate energy production. In the following a conventional system composed of a natural gas fired boiler (for thermal energy production) and a power plant (PP) connected to the electric grid (for electricity production) was considered. The boiler of the reference system was not dynamically simulated because its transient regime can be assumed as negligible.

The efficiency characteristics of the boiler were derived from manufacturer data of a commercially available product with 32 kW as nominal power [38]. A boiler with a nominal thermal output larger than the maximum thermal output of the proposed system (31.7 kW) was selected. The boiler manufacturer [38] suggests 0.911 as boiler efficiency for a part load ratio $P_{th,B}^{CS}/P_{th,B,max}^{CS}$ equal to 30% and 0.937 as boiler efficiency in the case of operation at rated output. In this study the boiler efficiency of the conventional

system $\eta_{B,CS}$ was evaluated by considering a linear interpolation of the manufacturer data:

$$\eta_B^{CS} = 0.911 + 0.001067 \times (P_{th,B}^{CS} - 10.44) \quad (5)$$

Exhaust and combustion losses rates were considered as negligible in this study. Therefore the primary power consumption of the boiler considered for the conventional system $P_{p,B}^{CS}$ is evaluated as follows:

$$P_{p,B}^{CS} = \frac{P_{th,B}^{CS}}{\eta_B^{CS}} \quad (6)$$

Concerning the reference electric grid efficiency η_{PP} , a figure of 0.461 was assumed: this value represents the power plant average efficiency in Italy, including transmission losses [29,47].

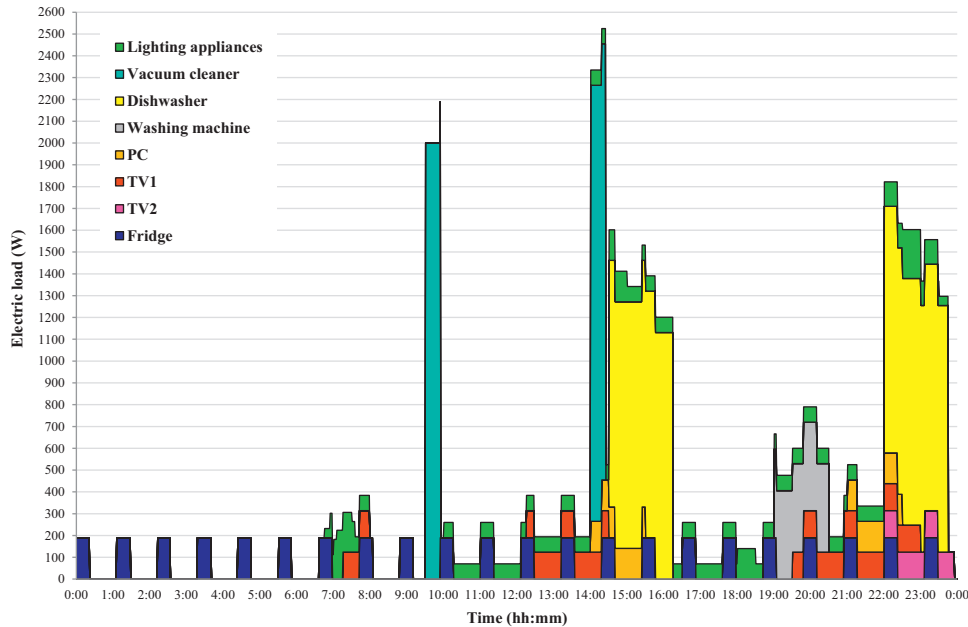


Fig. 7. Daily electric demand profile for a single flat.

Table 5

Results of sensitivity analysis carried out on tank volume.

	TANK1 (volume = 0.855 m ³)		TANK2 (volume = 0.738 m ³)		TANK3 (volume = 0.503 m ³)	
	Electric load-following operation	Thermal load-following operation	Electric load-following operation	Thermal load-following operation	Electric load-following operation	Thermal load-following operation
Hours of MCHP operation	2541	1639	2543	1765	2543	1686
$E_{el,MCHP}^{SP}$ (kJ)	2.47E+07	2.44E+07	2.49E+07	2.87E+07	2.50E+07	2.88E+07
$E_{th,MCHP}^{SP}$ (kJ)	8.08E+07	5.68E+07	8.14E+07	6.35E+07	8.10E+07	6.19E+07
$E_{p,MCHP}^{SP}$ (kJ)	1.32E+08	1.05E+08	1.33E+08	1.17E+08	1.33E+08	1.15E+08
$E_{th,B}^{SP}$ (kJ)	3.09E+07	1.20E+07	3.68E+07	1.88E+07	3.35E+07	1.42E+07
$E_{p,B}^{SP}$ (kJ)	3.34E+07	1.29E+07	3.97E+07	2.02E+07	3.62E+07	1.53E+07
$E_{el,grid}^{SP}$ (kJ)	1.88E+06	1.25E+07	1.92E+06	1.05E+07	1.92E+06	1.01E+07
$E_{p,grid}^{SP}$ (kJ)	4.07E+06	2.70E+07	4.16E+06	2.27E+07	4.17E+06	2.19E+07
$E_{p,TOT}^{SP}$ (kJ)	1.70E+08	1.45E+08	1.76E+08	1.60E+08	1.73E+08	1.52E+08

5. Results and discussions

The above-described scheme implemented in TRNSYS was simulated from November 15th to March 31st with 1-min simulation time step. The simulations were performed with two main aims:

- (1) evaluating the optimal value of storage volume able to minimize the primary energy consumption of the proposed system;
- (2) comparing the performance of the proposed system with the reference system. The comparison was performed in terms of primary energy consumption, equivalent carbon dioxide emissions and operating costs in order to estimate the suitability of the micro-cogeneration device-based plant.

The most important simulation results and the results of comparison between the proposed system and the conventional system are reported and commented in detail in the following sections.

5.1. Tank volume sensitivity analysis

A sensitivity analysis was carried out in order to estimate the influence of tank volume on the whole system performance; the operation with the three commercially available tanks described in Table 1 was evaluated under both electric and thermal load-following MCHP operation.

The main results of this analysis are reported in Table 5: the hours of MCHP operation, the electric energy produced by the cogeneration device $E_{el,MCHP}^{SP}$, the thermal energy supplied by cogenerator $E_{th,MCHP}^{SP}$, the primary energy consumed by the cogeneration unit $E_{p,MCHP}^{SP}$, the thermal energy produced by the boiler $E_{th,B}^{SP}$, the primary energy consumed by the boiler $E_{p,B}^{SP}$, the electric energy bought from the national grid $E_{el,grid}^{SP}$ and the primary energy $E_{p,grid}^{SP}$ consumed for producing $E_{el,grid}^{SP}$ (the value of $E_{p,grid}^{SP}$ was calculated by assuming 0.461 as efficiency of the Italian electric grid) derived by the simulations performed with each tank are specified. The total primary energy consumed by the whole system $E_{p,TOT}^{SP}$ (sum of $E_{p,MCHP}^{SP}$, $E_{p,B}^{SP}$ and $E_{p,grid}^{SP}$) was also reported in the last line of Table 5.

As it can be derived from this table, the tank with the largest volume (TANK1) allowed for minimizing the total primary energy consumption of the proposed system under both electric and thermal load-following control logic; whatever the MCHP control logic is, the tank with the intermediate volume (TANK2) was characterized by the larger total primary energy consumption, while the TANK3 produced an intermediate value of $E_{p,TOT}^{SP}$. In comparison to the operation with the TANK2, the TANK1 allowed for a percentage reduction of primary energy consumption of about 9.4% in the case of thermal load-following logic and of about 3.4% in the case of electric load-following operation mode; with regards to the

operation with the smaller hot water storage (TANK3), the TANK1 provided a percentage reduction of primary energy consumption of about 4.6% in the case of thermal load-following logic and of about 1.7% in the case of electric load-following operation mode; the percentage difference between TANK2 and TANK3 in terms of primary energy consumption ranges between 1.7% (in the case of electric load-following operation mode) and 5.0% (in the case of thermal load-following logic). Based on these results, the storage with the largest volume was considered as the best option for optimizing the plant performance. This is the reason why the following energy, environmental and economic analysis was carried out by considering the results achieved with the TANK1.

According to the data reported in Table 5, it can be also noticed that:

- whatever the tank volume is, electric load-following control strategy allowed for increasing the hours of MCHP operation in comparison to the heat load-following mode;
- the tank with the intermediate volume (TANK2) allowed for maximizing the hours of MCHP operation in comparison to the other two hot water storages in the case of heat-led operation;
- in the case of operation with the largest volume tank (TANK1), the cogeneration device provided 72% and 83% of the total thermal energy required for space heating and domestic hot water production under electric load-following logic and heat-load control mode, respectively; the remainder was supplied by the back-up boiler;
- in the case of operation with the largest volume tank (TANK1), the cogeneration device provided 93% and 66% of the total electric energy required by the building and the auxiliary systems (pumps, fan-coils, etc.) under electric load-following logic and heat-load control mode, respectively; the remainder was supplied by the central grid;
- during thermal load-following operation with the largest tank only 66% of the total electric energy consumption was met by the power from the MCHP unit: so in that case the mismatch between the cogeneration device power production and electricity consumption is quite relevant.

5.2. Energy analysis

As mentioned above, the proposed systems were compared with the reference systems in terms of primary energy consumption, carbon dioxide equivalent emissions and operating costs. The energy analysis is performed in this section. In order to compare the proposed system and the conventional system from an energy point of view, the following indicator ΔE_p is adopted:

$$\Delta E_p = \frac{(E_{p,TOT}^{CS} - E_{p,TOT}^{SP})}{E_{p,TOT}^{CS}} \quad (7)$$

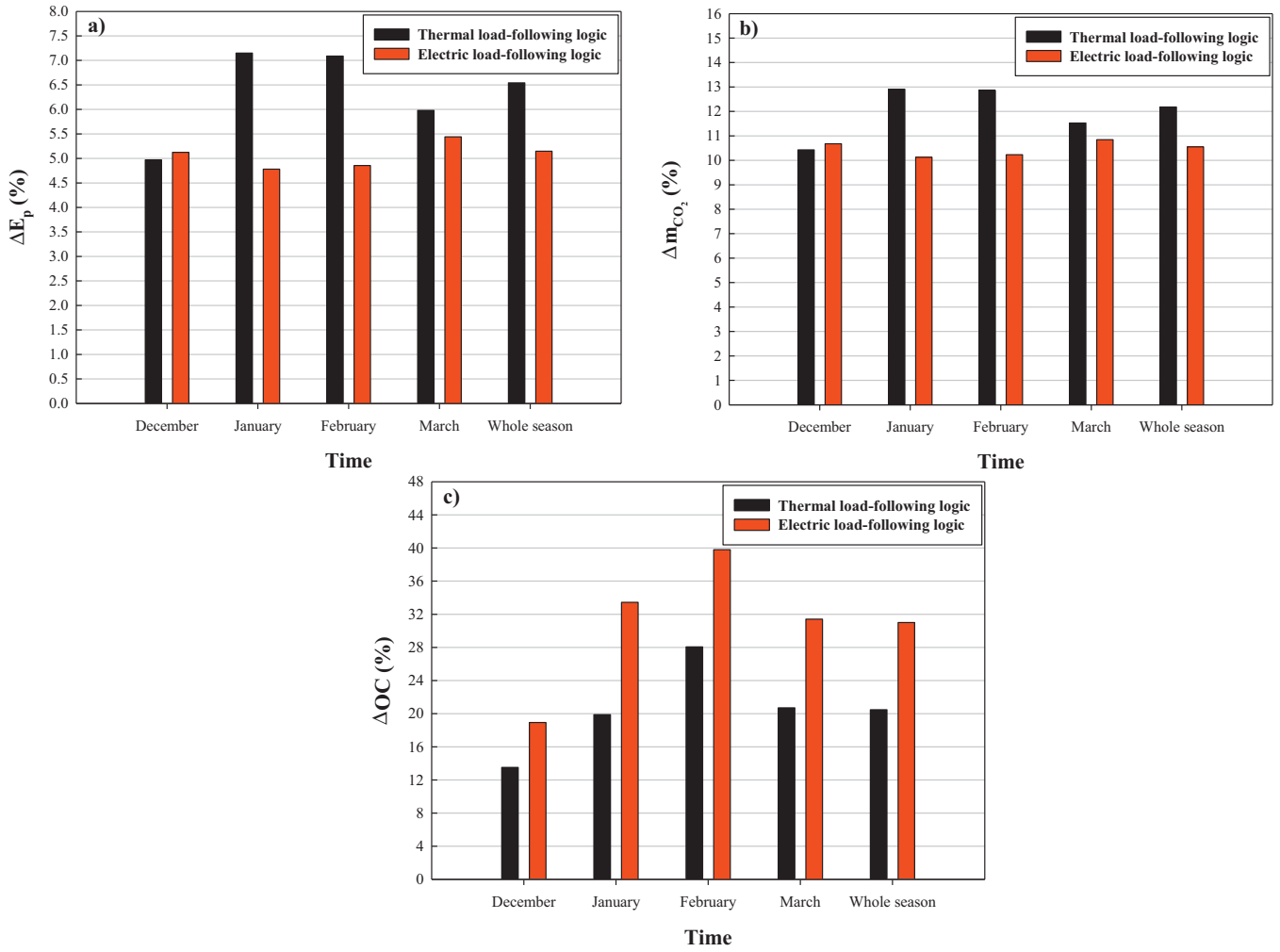


Fig. 8. Results of comparison between the proposed system and the conventional system in terms of primary energy consumption (a), carbon dioxide equivalent emissions (b), and operating cost (c).

where $E_{p,TOT}^{SP}$ is the primary energy consumed by the proposed system and $E_{p,TOT}^{CS}$ is the primary energy consumed by the conventional system for supplying the same output ($E_{el,MCHP}^{SP} + E_{th,MCHP}^{SP} + E_{th,B}^{SP} + E_{el,buy}^{SP}$) of the proposed system.

The values of $E_{p,TOT}^{SP}$ and $E_{p,TOT}^{CS}$ were computed as reported below:

$$E_{p,TOT}^{SP} = \frac{E_{p,MCHP}^{SP} + E_{th,B}^{SP}}{\eta_B^{SP}} + \frac{E_{el,buy}^{SP}}{\eta_{PP}} \quad (8)$$

$$E_{p,TOT}^{CS} = \frac{E_{th,MCHP}^{SP}}{\eta_B^{CS}} + \frac{E_{th,B}^{SP}}{\eta_B^{CS}} + \frac{E_{el,MCHP}^{SP}}{\eta_{PP}} + \frac{E_{el,buy}^{SP}}{\eta_{PP}} \quad (9)$$

where

- the values of $E_{p,MCHP}^{SP}$, $E_{th,B}^{SP}$, $E_{el,buy}^{SP}$, $E_{th,MCHP}^{SP}$, $E_{el,MCHP}^{SP}$ were derived by TRNSYS simulations;
- the values of η_B^{SP} and η_B^{CS} represent, respectively, the boiler efficiency of the proposed system (calculated by using the Eq. (3)) and the boiler efficiency of the conventional system (calculated by using the Eq. (5));
- η_{PP} is the power plant average efficiency in Italy, including transmission losses [29,47] (assumed equal to 0.461).

In Fig. 8a the values of ΔE_p calculated under both electric and thermal load-following control logic are reported with reference to the operation during each single month and during whole heating season. According to Eq. (7), a positive value of ΔE_p indicates that the proposed system allows for a primary energy saving in comparison to the conventional system. Fig. 8a shows that, compared to the conventional system, the proposed system allows for a primary energy reduction during the whole heating season of about 6.5% in the case of thermal load-following operation and 5.1% under electric load-following control strategy. In addition this figure highlights that: (i) also during each single month operation the values of ΔE_p are positive, and (ii) the thermal load-following operation appears to be more convenient than the electric load-following operation from an energy point of view (except during December).

5.3. Environmental analysis

The choice of the energy conversion technology cannot be based only on the energy performance, but it should be also affected by the assessment of the environmental impact. The assessment of pollutant emissions was also performed in this study through an energy output-based emission factor approach [48]. According to this approach, the mass m_x of a given pollutant x emitted while producing the energy output E can be worked out as:

$$m_x = u_x^E \cdot E \quad (10)$$

where u_x^E is the energy output-based emission factor, that is, the specific emissions of x per unit of E . This factor depends upon several operating and structural variables, such as the specific equipment, partial load operation, age, state of maintenance, outdoor conditions, pollutant abatement systems, and so forth.

Eq. (10) can be used for considering the emissions of any type of GHG for any system. However, as far as emissions from fuel burning are concerned, in general CO₂ emissions prove to be quantitatively much more important than other GHG emissions [49]. Therefore, at first approximation, GHG emissions other than CO₂ can be neglected. Thus, only the carbon dioxide emissions will be considered in the analyses carried out in this paper. For a given fuel, the CO₂ emission factor referred to the primary energy released when burning the fuel can be considered constant at first approximation. In fact it can actually change when the unit operates in generic off-design conditions. In this case, typically the combustion characteristics worsen, bringing about incomplete combustion and thus a decrease of CO₂ specific emissions [50,51]. However, this behavior can be neglected at first approximation, as done in several studies [50,52,53]. According to the values suggested in [29,54,55], in this study the CO₂ emission factor associated to the natural gas consumption $u_{CO_2}^{E_p}$ is assumed equal to 200 gCO₂/kWh_p: this value represents the equivalent CO₂ emissions for a primary energy unit consumed by burning out natural gas.

The average CO₂ emissions for producing electricity depend upon the technology mix adopted. Thus, the emissions are significantly lower where the generation system is based upon renewable sources such as hydroelectric power. Italy has a power generation system with no nuclear power and with about 20% of hydroelectricity; the main part is thermal-based, with a mix of fuels from coal to oil. According to the values suggested in [54,55], in this paper the CO₂ emission factor for electricity production $u_{CO_2}^{E_{el}}$ is assumed equal to 525 gCO₂/kWh_{el}: this value represents the equivalent CO₂ emissions for an electric energy unit consumed.

In the following the equivalent carbon dioxide emissions of the proposed system are estimated and compared to those of the conventional system. This comparison was performed by using the following equation:

$$\Delta m_{CO_2} = \frac{(m_{CO_2}^{CS} - m_{CO_2}^{SP})}{m_{CO_2}^{CS}} \quad (11)$$

where

$$m_{CO_2}^{SP} = u_{CO_2}^{E_p} \cdot E_{p,MCHP}^{SP} + u_{CO_2}^{E_{th,B}} \cdot \frac{E_{th,B}^{SP}}{\eta_B^{SP}} + u_{CO_2}^{E_{el}} \cdot E_{el,buy}^{SP} \quad (12)$$

$$m_{CO_2}^{CS} = u_{CO_2}^{E_p} \cdot \frac{(E_{th,MCHP}^{SP} + E_{th,B}^{SP})}{\eta_B^{CS}} + u_{CO_2}^{E_{el}} \cdot (E_{el,MCHP}^{SP} + E_{el,buy}^{SP}) \quad (13)$$

In Fig. 8b the values of Δm_{CO_2} calculated under both electric load- and thermal load-following logic are reported with reference to the operation during each single month and during whole heating season. According to the Eq. (11), a positive value of Δm_{CO_2} indicates that the proposed system allows for a reduction of carbon dioxide equivalent emissions in comparison to the conventional system. Fig. 8b shows that, compared to the conventional system, the proposed system allows for a reduction of equivalent carbon dioxide emissions during the whole heating season of about 12.2% in the case of thermal load-following control logic and 10.6% under electric load-following operation. In addition this figure highlights that: (i) also during each single month operation the values of Δm_{CO_2} are positive and (ii) the thermal load-following operation appears to be more convenient than the electric load-following operation from an environmental point of view (except during December). Finally it can be noticed that the values of Δm_{CO_2}

showed in Fig. 8b are about two times larger than the values of ΔE_p reported in Fig. 8a: this is mostly due to fact that the CO₂ emission factor for electricity production is about 2.6 times greater than the CO₂ emission factor associated to the natural gas consumption.

5.4. Economic analysis

Economic assessment of the proposed cogeneration-based plant requires comparison of its cost to that of the reference system. In this paper the operating costs due to both natural gas and electric energy consumptions were evaluated in detail; the revenue from selling the electric energy surplus has also been taken into account and subtracted from the operating costs. Both the investment and maintenance costs were not considered because the simulations were limited to the sole heating season (during the period from November 15th to March 31st) instead of the whole year; therefore it was not possible to evaluate the number of years required to recover the usual higher investment and maintenance costs of the proposed system with regards to the conventional system.

The operating costs OC of both the proposed system SP and the conventional system CS, respectively, were calculated by using the following formulas:

$$OC_{SP} = CU_{ng} \cdot \frac{(E_{p,MCHP} + E_{th,B}^{SP}/\eta_B^{SP})}{(3600 \cdot LHV_{ng} \cdot \rho_{ng})} + CU_{el,buy} \cdot E_{el,buy}^{SP} - CU_{el,sell} \cdot E_{el,sell}^{SP} \quad (14)$$

$$OC_{CS} = CU_{ng} \cdot \left(\frac{E_{th,B}^{SP}/\eta_B^{CS}}{(3600 \cdot LHV_{ng} \cdot \rho_{ng})} \right) + CU_{el,buy} \cdot E_{el,MCHP} \quad (15)$$

where

- CU_{ng} is the unit cost of natural gas;
- LHV_{ng} is the lower heating value of natural gas (assumed equal to 49,599 kJ/kg);
- ρ_{ng} is the density of natural gas (assumed equal to 0.72 kg/m³);
- $E_{p,MCHP}$ is the primary energy consumed by the cogeneration device provided by the MCHP Annex 42 model [25,26];
- $E_{th,B}^{SP}$ is the thermal energy produced by the natural gas-fired boiler of the proposed system;
- η_B^{SP} is the efficiency of natural gas-fired boiler of the proposed system (calculated by means of Eq. (3));
- $CU_{el,buy}$ is the unit cost of the electric energy bought from the national central grid;
- $E_{el,buy}^{SP}$ is the electric energy bought from the national central grid by the proposed system;
- $CU_{el,sell}$ is the unit cost of the electric energy sold to the national central grid;
- $E_{el,sell}^{SP}$ is the electric energy sold to the national central grid by the proposed system;
- η_B^{CS} is the efficiency of natural gas-fired boiler composing the conventional system (calculated by means of Eq. (5));
- $E_{el,MCHP}$ is the electric energy produced by the cogeneration device.

According to the Italian scenario [56], the unit cost of natural gas CU_{ng} for residential applications is the sum of five terms:

- a variable rate, depending on both the region of Italy where the natural gas is consumed and the level of natural gas consumption;
- the regional tax, depending on both the region of Italy where the natural gas is consumed as well as the level of cumulated natural gas consumption;

- the excise tax, that is lower for cogenerative use in comparison to applications other than combined heat and power production. This difference is due to the fact that, taking into account that a micro-cogeneration system would require a higher investment for the user as compared to the conventional separate production of heating and electricity, some incentives were adopted by the Italian government for MCHP units to be financially feasible. In the case of applications other than combined heat and power production the excise tax depends on both the region of Italy where the natural gas is consumed and the level of cumulated natural gas consumption, while it is constant (0.0001348 €/Sm³) when cogenerative use is considered;
- the value added tax (VAT), that depends on the level of natural gas consumption (two consumption bands are considered);
- the yearly fixed charge, that depends on the region of Italy where the city is located.

Table 6 shows both the different rates composing the natural gas unit cost and the total unit cost of natural gas for domestic clients in Naples; a yearly fixed charge of about 91 € associated to the domestic clients in Naples must be added to the operating cost reported in Table 6.

Regarding the unit cost of electric energy purchased from the grid, it is worth noting that in Italy the residential consumers have time-of-use rates for the electricity prices. The values of $CU_{el, buy}$ vary depending on [57]:

- the day (the cost during a weekend day is lower with respect to a week day);
- the hour of the day;
- the level of cumulated electric energy consumption.

According to Italian Law [58], the unit cost of the electric energy sold to the national central grid $CU_{el, sell}$ depends only on: (i) the year, (ii) the city, (iii) the day (week day, Saturday and Sunday are differentiated), as well as (iv) the hour of the day. In this paper the values of $CU_{el, sell}$ referred to the year 2012 were considered.

Fig. 9a shows the daily profile of the unit cost of the electric energy bought from the national central grid $CU_{el, buy}$ during a week day for each of the four different electric energy consumption bands; the values of $CU_{el, buy}$ during a weekend day do not change with the hour of the day, but they depend only on the electric energy consumption as follows:

- from 0 to 1800 kWh: $CU_{el, buy} = 0.225209 \text{ €/kWh}$;
- from 1801 to 2640 kWh: $CU_{el, buy} = 0.242479 \text{ €/kWh}$;
- from 2641 to 4400 kWh: $CU_{el, buy} = 0.283674 \text{ €/kWh}$;
- higher than 4400 kWh: $CU_{el, buy} = 0.326354 \text{ €/kWh}$.

Fig. 9b reports the values of $CU_{el, sell}$ as a function of the time during a week day, Saturday and Sunday.

As mentioned above, the evaluation of the economic performance indices is also necessary to complete the comparison between the proposed and conventional systems. The values of OC_{SP} and OC_{CS} were compared by means of the following parameter:

$$\Delta OC = \frac{OC_{CS} - OC_{SP}}{OC_{CS}} \quad (16)$$

In Fig. 8c the values of ΔOC calculated under both electric and thermal load-following logic are reported with reference to the operation during each single month and during the whole heating season. According to Eq. (16), a positive value of ΔOC indicates that the proposed system allows for a reduction of operating costs in comparison to the conventional system. Fig. 8c shows that, compared to the conventional system, the proposed system allows for

Table 6
Natural gas unit cost for domestic clients in Naples.

	Band consumption (Sm ³ /year)	From 121 to 480	From 481 to 1560	From 1561 to 5000	From 5001 to 80,000	From 80,001 to 200,000
Variable rate (€/Sm ³)	From 0 to 120	0.479882	0.627644	0.623244	0.586119	0.531819
Regional tax (€/Sm ³)		0.019	0.031	0.031	0.031	0.031
Excise tax for cogenerative use (€/Sm ³)						
Excise tax for applications other than cogeneration (€/Sm ³)				0.0001348		
VAT(%)		10	21	21	21	21
Natural gas unit cost for cogenerative use (€/Sm ³)		0.5489185	0.7971223	0.7917983	0.7468771	0.6811741
Natural gas unit cost for applications other than cogeneration (€/Sm ³)		0.5905702	0.9421592	0.9731352	0.928214	0.862511

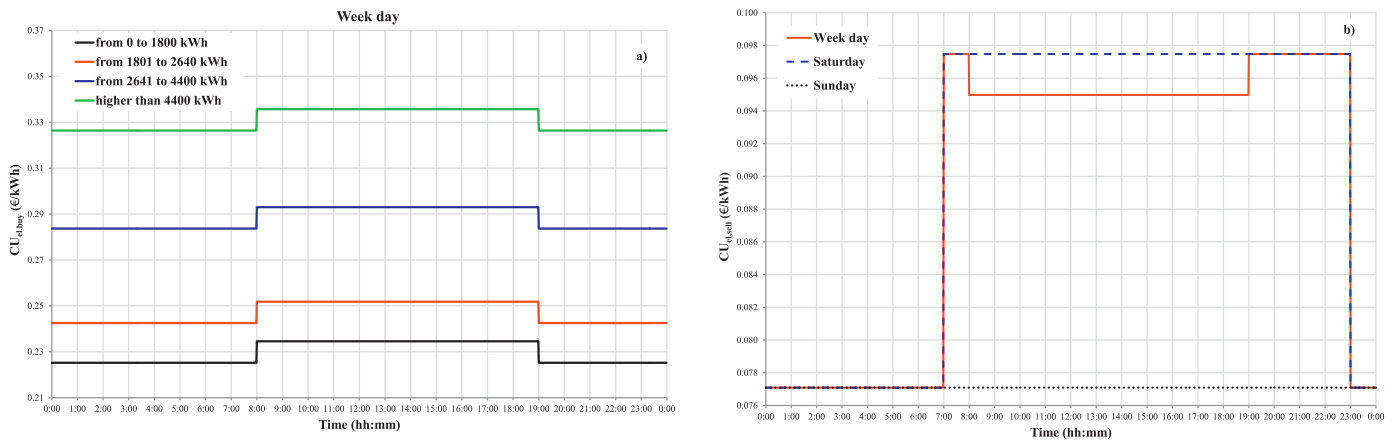


Fig. 9. Daily unit cost profile of electric energy bought from the grid during a week day (a) and sold to the grid during the week (b).

a reduction of operating costs during the whole heating season of about 20.5% in the case of thermal load-following logic and 31.0% under electric load-following control strategy. In addition, this figure highlights that: (i) also during each single month operation the values of ΔOC are positive and (ii) the electric load-following operation is more convenient than the thermal load-following operation from an operating cost point of view (this last result is different from the conclusions achieved by comparing the proposed and conventional systems from both energy and environmental point of views).

6. Conclusions

In this paper the performance of natural gas-fuelled building-integrated micro-cogeneration system was simulated by using the whole building simulation software TRNSYS and compared with those of a conventional system (composed of a natural gas fired boiler and central electric grid) in terms of primary energy consumption, carbon dioxide equivalent emissions and operating costs in order to assess the potential energy and emissions saving attributable to the MCHP system. Simulations were performed from November 15th to March 31st with 1-min simulation time step, by considering a multi-family house located in Naples (south of Italy) compliant with the Italian Standards in terms of the transmittance values of both the walls and windows.

The transient nature of building and occupant driven loads, the part-load characteristics of the cogeneration system, the interaction between the loads and the system output, and system energy management and control were considered in the analysis

The simulations results showed that, in comparison to the system assumed as a reference, the proposed system allows for a significant reduction of primary energy consumption, carbon dioxide emissions and operating costs. In comparison to the electric load-following logic, the heat demand-following control strategy of the MCHP unit appeared more convenient from both energy and emissions point of views, but less advantageous in terms of operating costs. In more particular, the study revealed that, with reference to the Italian electricity grid mix, the MCHP thermal load-following approach can offer a primary energy demand saving of about 6.5%, a carbon dioxide equivalent emissions reduction of about 12.2%, and an operating cost lowered of about 20.5%; under MCHP electric load-following operation, these percentage differences relative to the conventional mean of supplying heat and power becomes 5.1%, 10.6% and 31.0%, respectively.

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